Dynamic characterization of wind turbine gearboxes using Order-Based Modal Analysis

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Abstract

This paper describes an extensive measurement campaign in order to characterize gearbox vibro-acoustic behavior. The measurements have been performed at ZF Wind Power on a 13.2MW test rig facility. Accelerations have been measured at more than 250 locations on the test rig and for different load levels and operating conditions.

It is important to extract from the test rig measurements those parameters which are representative for the gearbox dynamic behavior. Several techniques, such as Operational Modal Analysis (OMA) and Order Based Modal Analysis (OBMA), have been applied to the acceleration data in order to extract the modal parameters in the test rig configuration. While the first technique shows some limitations, such as the so-called "end-of-order" effect, the second technique combining advanced Order Tracking with Operational Modal Analysis, does not suffer from "end-of-order" related peaks in the spectrum and identifies only the physical poles of the system.

1 Introduction

In the past years, the number of wind farms has increased and some of them were installed quite close to residential areas. This is one of the main reasons for which noise emission levels are a matter of public interest. The airborne rotor noise had a drastic reduction based on improved blade design, new control strategies and tip speed limitations. However, mechanical noise resulting from gearbox, generator etc. - primarily tonal of nature - can become a new sound challenge. In the case of the gearbox, the sound originates from gear excitations which are transmitted through shafts, bearings and the housing [1].

The gearbox is one of the key subsystems in a geared wind turbine providing the task to transfer power from the low speed shaft connected to the rotor to the high speed shaft connected to the generator. A deep knowledge into gearbox dynamics becomes of fundamental importance and noise and vibration measurements are demanded [2].

The demanded measurements are mainly quality estimation methods for gear mesh vibrations and overall sound power levels. They are based on standard techniques for the estimation of dynamic characteristic in general applications and are not focusing on the wind turbine gearbox case. Building on existing techniques such as “Order Tracking” and “Operational Modal Analysis”, a dedicated methodology for the analysis of operational gearbox dynamic behavior is under development. Particular attention is given to the separation between structural resonances and excitation orders.

Operational Modal Analysis (OMA) is used to derive an experimental modal model from vibration measurements in operational conditions. In this case, one of the main assumptions on which the method is based is violated: the input excitation, self-induced gear excitation, is not a broadband white noise, but there are several rpm dependent frequencies (gear meshing orders). This means that OMA cannot be applied in a straightforward way. These gear excitation frequencies with high vibration levels can be wrongly considered as resonance frequencies of the system. In order to face these problems in such a challenging case, new procedures need to be developed and validated.

The main objective of the work is to identify the technique that allows a better understanding of the field behavior of a gearbox by means of test rig measurements before the installation on the wind turbine takes place.

2 Theoretical background

2.1 Operational Modal Analysis (OMA)

The OMA technique, also known as output-only modal analysis, allows to identify modal parameters by using operational measurement such as accelerations measured on several points on the structure [4]. The OMA technique is applied when the input forces cannot easily be measured. The system has to comply with some assumptions: must be linear time invariant and the excitation forces must have a flat white noise spectrum in the frequency band of interest. The better these assumptions are fulfilled, the better and more reliable the estimated modal parameters [5]. The identification technique is similar to the classical input-output modal analysis with the substantial difference that, instead of impulse and frequency responses, it uses auto- and cross-correlation and auto- and cross-powers between signals measured simultaneously at different locations. So, there is the need to identify several reference signals that should be as less noisy as possible and that should be able to identify as many modes as possible. Operational Polymax [6] method and Stochastic Subspace Identification by using the Balanced Realization (BR) technique have been applied to the gearbox data for identifying natural frequencies, damping ratios and mode shapes.

2.1.1 Operational Polymax method

Frequency-domain Operational Modal Analysis methods, such as Polymax, require output spectra as primary data. Under the assumption of white noise spectrum excitation, output spectra can be modeled in a very similar way as Frequency Response Functions (FRFs). The modal decomposition of an FRF matrix $H(\omega)$ is:

$$H(\omega) = \sum_{i=1}^{n} \frac{\{v_i\} < l_i^T >}{j\omega - \lambda_i} + \frac{\{v_i^*\} < l_i^H >}{j\omega - \lambda_i^*} \tag{1}$$

where n is the number of complex conjugated mode pairs, * is the complex conjugate operator, ^{H} is the complex conjugate transpose (Hermitian) of a matrix, $\{v_i\}$ are the mode shapes, $<l_i^T>$ are the modal participation factors and λ_i are the poles. The relationship that occurs between poles, eigenfrequencies ω_i and damping ratios ξ_i is:

$$\lambda_i \lambda_i^* = -\xi_i \omega_i \pm j\sqrt{1-\xi_i^2}\,\omega_i \tag{2}$$

The input spectra $[S_{uu}(\omega)]$ and output spectra $[S_{yy}(\omega)]$ of a system represented by the FRF matrix are related as:

$$\left[S_{yy}(\omega)\right]=\left[H(\omega)\right]\left[S_{uu}(\omega)\right]\left[H(\omega)\right]^{H} \tag{3}$$

In case of operational data, output spectra are the only available information. The deterministic knowledge of the input is replaced by the assumption that the input is white noise. In fact, the most important property of white noise is that it has a constant power spectrum. Hence $[S_{uu}]$ is independent of the frequency ω. Combining Equation (1) and (3), the modal decomposition of the output spectrum matrix is obtained as shown in Equation (4):

$$\left[S_{yy}(\omega)\right]=\sum_{i=1}^{n}\frac{\{v_i\}<g_i>}{j\omega-\lambda_i}+\frac{\{v_i^*\}<g_i^*>}{j\omega-\lambda_i^*}+\frac{\{g_i\}<v_i>}{-j\omega-\lambda_i}+\frac{\{g_i^*\}<v_i^*>}{-j\omega-\lambda_i^*} \tag{4}$$

where $<g_i>$ are the so-called operational reference factors, which replace the modal participation factors in case of output-only data. The goal of OMA is the identification of the right hand side terms of Equation (4) based on the measured output data pre-processed into output spectra.

2.1.2 Stochastic Subspace Identification – Balanced Realization

Another advanced OMA method is the Stochastic Subspace Identification (SSI) one. The term subspace means that the method identifies a state-space model and that it involves a Singular Value Decomposition (SVD) truncation step. There are several variants in this class of methods including Canonical Variate Analysis (CVA) and Balanced Realization (BR).

In this method a so-called stochastic state space model is identified from output correlations or directly from measured output data [7]. The state space model is written in Equation (5):

$$\begin{aligned} x_{k+1} &= Ax_k + w_k \\ y_k &= Cx_k + v_k \end{aligned} \tag{5}$$

where y_k is the sampled output vector, x_k is the discrete state vector, w_k is the process noise due to disturbances and unknown excitation of the structure, v_k is the measurement noise, mainly due to sensor inaccuracy, but also to the unknown excitation of the structure; k is the time instant. The matrix A is the state transition matrix that completely describes the dynamics of the system by its eigenvalues; C is the output matrix. The only known terms in Equation (5) are the output measurements y_k and the challenge is to determine the system matrices A and C from which the modal parameters can be derived.

The derivation of the modal parameters starts with the eigenvalue decomposition of A:

$$A=\Psi\Lambda_d\Psi^{-1} \tag{6}$$

where Ψ is the eigenvector matrix and Λ_d is a diagonal matrix containing the discrete-time eigenvalues μ_i which are related to the continuous-time eigenvalues λ_i as:

$$\mu_i=\exp(\lambda_i\Delta t) \tag{7}$$

The eigenfrequencies and damping ratios are related to λ_i as expressed in Equation (2).Finally, the mode shapes are found as:

$$V=C\Psi \tag{8}$$

2.2 Order-Based Modal Analysis (OBMA)

In the presence of rotating machinery (gearboxes, engines, pumps, etc.), the so-called “orders” are identified as multiples or fractions of the rotation speed. They are the proportional constants between the rotation speed and the frequency. The “Order Tracking” theory estimates the amplitude and the phase of the orders for varying rotation speed.

During a run-up or run-down test, the measured response is mainly caused by rotational excitation. This is the main reason behind the idea to perform OMA on tracked orders rather than on the overall spectrum. The method considers the run-up as a multi-sine sweep excitation and combines advanced order tracking techniques with operational modal analysis to identify the modal parameters.

2.2.1 Order tracking

Traditionally, two methods have been employed to digitally track orders which results from rotating components in noise and vibration problems. These two methods are the Fast Fourier Transform (FFT) based method [8] and the angular re-sampling based method [9]. Recently, a Kalman filter approach has been introduced to track orders in noise and vibration data [10].

Another important method, known as Time variant Discrete Fourier Transform (TVDFT), has been developed as an alternative order tracking method. The TVDFT method is a combination of the FFT and the re-sampling based methods. It has many of the advantages of the re-sampling based order tracking, while reducing the computational time. It performs much better than the classical FFT-based approach, which gives unreliable phase estimates.

The TVDFT is based upon a discrete Fourier transform which has a kernel whose frequency varies as a function of time defined by the rpm of the machine [11]. This kernel is a cosine or sine function of unity amplitude with an instantaneous frequency matching the one of the tracked order at each instant in time. It can be formulated as represented in equations (9) (10), but also in a complex exponential format similar to the corresponding Fourier transform.

$$a_n = \frac{1}{N}\sum_{n=1}^{N} x(n\Delta t)\cos(2\pi \int_0^{n\Delta t}(o_n \bullet \Delta t \bullet rpm/60)dt) \tag{9}$$

$$b_n = \frac{1}{N}\sum_{n=1}^{N} x(n\Delta t)\sin(2\pi \int_0^{n\Delta t}(o_n \bullet \Delta t \bullet rpm/60)dt) \tag{10}$$

where o_n is the order which is being analyzed, a_n and b_n are the Fourier coefficients of the cosine and sine terms for o_n, $x(n\Delta t)$ are the data samples, N is the number of data samples within an analysis block.

This transform is best suited to estimate orders with a constant order bandwidth. The instantaneous rpm of the machine is integrated to obtain the number of revolutions the shaft has rotated through at each instant in time, as done in the re-sampling algorithm. Constant order bandwidth estimates can be obtained by performing the transform over the number N of time points required to achieve the desired order resolution Δo. As the rpm increases, the angular spacing of the re-sampled samples $\Delta\theta$ increases and the transform is applied over a shorter time, as can be seen from Equation (11):

$$\Delta o = \frac{1}{N * \Delta\theta} \tag{11}$$

This transform is performed for the desired orders and not for the full spectrum as done in the others order tracking methods. In the following sections the technique is applied to the run up data to accurately track the orders in both amplitude and phase.

2.2.2 OMA on tracked orders

It is possible to investigate the relationship between the modal model and an order by assuming the structure to be excited by a rotating mass with increasing frequency [12]. It can be represented by two perpendicular forces of equal amplitude and in quadrature (with 90° phase difference). The measured response $Y(\omega)$ in the frequency domain is:

$$Y(\omega) = H_{(:,fx)}(\omega)F_x(\omega) + H_{(:,fy)}(\omega)F_y(\omega) \tag{12}$$

where F represents the force and H the corresponding column of the transfer function matrix. Taking into account the relation between the two correlated forces, the response of the structure can be written as (considering only the positive frequency axis):

$$Y(\omega) \propto \omega_0^2 (H_{(:,fx)}(\omega) - jH_{(:,fy)}(\omega))\delta(\omega - \omega_0) \tag{13}$$

where ω_0 is the rotation speed. From this equation, it is clear that the measured output is proportional to the squared rotation speed and to a complex combination of two structural FRFs related to x and y excitation. A structural FRF can be decomposed in a modal sense as:

$$H_{(:,\bullet)}(\omega) = V(j\omega I - \Lambda)^{-1} L_{\bullet} + \frac{1}{\omega^2} LR_{\bullet} + UR_{\bullet} \tag{14}$$

where V, Λ, L are the complex-valued modal parameters (respectively, the mode shape matrix, the diagonal matrix containing the complex poles and the modal participation matrix). $LR_{\bullet}$ and $UR_{\bullet}$ are the real-valued lower and upper residuals, modeling the influences of the modes outside the considered frequency band. Finally, combining equation (12) and (13) we obtain:

$$Y(\omega) \propto \omega^2 (V(j\omega I - \Lambda)^{-1}(L_x - jL_y) + \frac{1}{\omega^2}(LR_x - jLR_y) + (UR_x - jUR_y) \tag{15}$$

The equation (15) shows that modal analysis can be applied to displacement orders taking into account the following observations:

- Displacement orders are proportional to the squared rotating speed and, therefore, acceleration orders are proportional to the 4th power of the rotating speed. In classical modal analysis, acceleration FRFs are proportional to the squared frequency axis.
- Complex upper and lower residuals, while in classical modal analysis they are real.
- Complex participation factor; also in classical modal analysis they can be complex.

It can be concluded that methods like Polymax [5] that estimates a right matrix-fraction model are robust against these observations, but methods working on the pole-residue formulation like the LSFD (Least Squares Frequency Domain) method need a new formulation in order to be applied.

3 Measurement campaign

3.1 Experimental setup

The measurement campaign took place on the 13.2 MW dynamic test rig at ZF Wind Power in Lommel, Belgium. Two prototype gearboxes were placed in a back-to-back configuration. This configuration allows limiting the energy requirements to operate the test rig, as the energy used to drive the gearboxes in the motor is recovered by the generator and fed back into the system. Thus, one gearbox (3.2 MW, named P3) operates in the so-called “generator mode” as in the wind turbine, while the second one (3 MW, named P2) is in the “motor mode”. They have slightly different gear ratio, which means that the motor mode gearbox does not run at its nominal speed when the generator mode one is being tested. The gearboxes can be tested under representative loading conditions using parameterized load cases that can be programmed into the test rig controller. The tested gearbox (P3) is driven at a certain time varying speed corresponding to variable wind speeds and loaded with a certain time varying torque by the generator corresponding to the loading from the grid. It experiences test conditions very similar to the ones that it would experience in the wind turbine in operating conditions.

More than 250 points were measured by using tri-axial accelerometers [13]. Several operating conditions were taken into account during the measurements: standstill conditions, run-ups at different torque levels, constant speed at different torque levels for two different speeds. Because only 37 tri-axial accelerometers were available 7 different batches sweeping all measurement points had to be performed.

For each batch, ten different operating conditions were measured and, at the end, all the accelerometers were moved to new positions and the work program was repeated once more. A global overview of the sensor locations both in the finite element (FE) model and in the experimental model are shown in Figure 1. The points were defined under different groups. Each group represents a different component of the gearbox. A list of the number of points divided by groups is shown in Table 1.

The X global axis goes from the Low Speed Shaft (LSS) to the High Speed Shaft (HSS) of the main tested gearbox (P3) that means from P2 to P3 in Fig.3; Z axis point vertically up, while Y axis is defined to get a consistent axis system.

Component	**Abbreviation**	**Number of measurement locations**
Tested gearbox (gearbox 1) = P3		
Torque Arm Cover	TAC	12
Torque Arm	TA	30
Low Speed Stage Ring Wheel	LSRW	12
Intermediate Bearing Housing	IBH	24
Intermediate Speed Stage Ring Wheel	ISRW	12
Bearing Housing	BH	36
High Speed Stage Housing	HSH	76
Torque Arm Cover	TAC	12
Counter gearbox (gearbox 2) = P2		
	P2	27
Cassette + Motor 1 (tested gearbox, P3) + Motor 2 (counter gearbox, P2)		
	CASS	27
Total		256

Table 1: Measurement point list

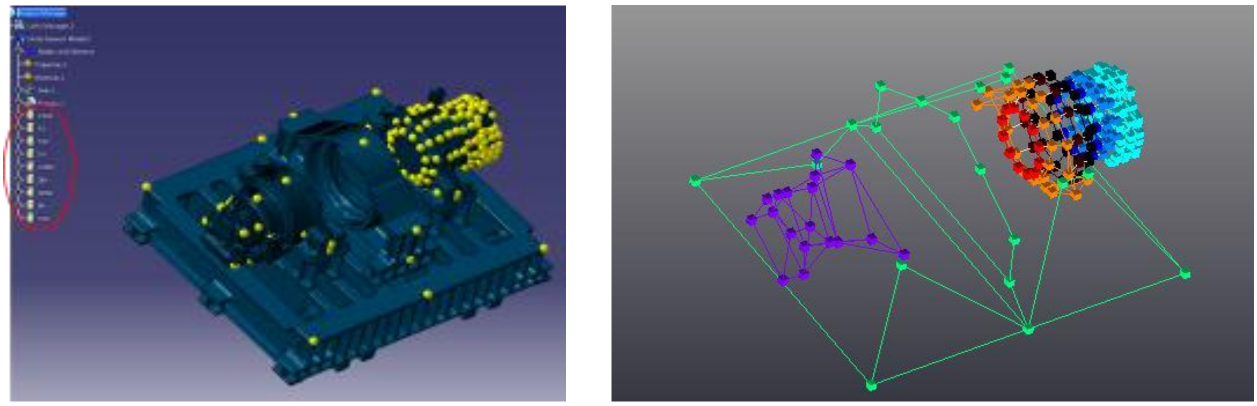

Figure 1: Test rig FE model and measurement points

3.2 Experimental data analysis

3.2.1 OMA: "end-of-order" effect

In some applications, the "natural" flat spectrum excitation provided by the rotation of the shafts inside the gearbox can be used as source for the application of OMA. In order to fulfill the white noise spectrum hypothesis, the run up case in which the gearbox is running through its operational rotation speed range is considered. In fact, the harmonics (orders), related to the number of revolutions, are sweeping through a broad frequency band and they are useful excitation for estimating the modal parameters. The gearbox is rotating from 200 rpm to 1500 rpm, with a speed run up rate equal to 5 rpm/s. During the measurements, the same run up case was performed at three different torque levels (33%, 66% and 100%).

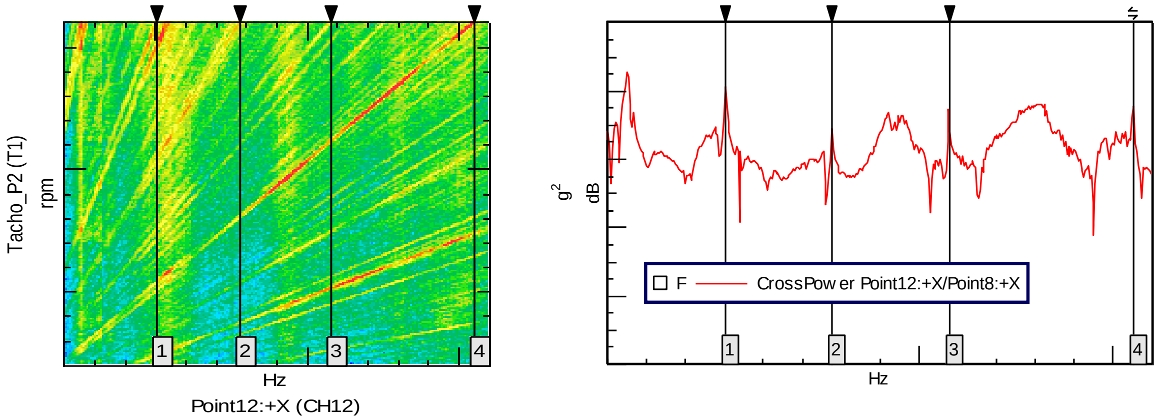

Figure 2: Rpm-frequency spectrogram of the gearbox run up in a certain frequency range and corresponding overall spectrum

Accelerations were collected by means of tri-axial accelerometers at a quite high sampling frequency (16384 Hz) requiring a down sampling to 1500 Hz in order to put focus on the frequency band of interest: 0 – 500 Hz. While the most part of accelerometers were moved between each batch to cover all the measurement points, 8 accelerometers were kept at the same position to be used as reference channels for the calculation of the cross-powers. Two points (named point 5 and point 8) are considered as the most suitable reference channels since their spectra have a quite good repeatability between the different batches. The smaller the differences, the better the estimation of the modal parameters is. To perform Operational Modal Analysis, a pre-processing is necessary to convert time data to auto- and cross-powers. First of all, auto- and cross-correlation functions are calculated and an exponential window is usually required before computing the FFT. The exponential window reduces the effect of leakage and the influence of the high time lags, which have a large variance.

Operational Polymax has been applied to estimate the modal parameters. In a previous work [13] the modal parameters were estimated at varying torque level. For increasing values of the applied torque, a small shift toward higher natural frequencies can be seen. The modal analysis can be performed in two different ways. The first approach, the so-called multi-run approach [14], considers the different batches separately obtaining partial mode shapes. Afterwards, these partial mode shapes are combined by means of the multi-run analysis to get the global mode shapes. These partial mode shapes are scaled with respect to the common degrees of freedom and the poles are evaluated as averaged poles between those obtained by the single batches. The second approach allows analyzing the data by considering all the batches together since, the repeatability of the measurements of the sensors in common between the different runs is quite good.

At first sight, the results of the Operational Polymax seem to be very satisfactory, but they need to be interpreted with care. Figure 2 shows the overall spectrum used to extract the modes as well as the rpm-frequency spectrogram. A comparison of the two graphs reveals that some of the peaks in the overall spectrum (at a first sight the sharpest ones) are originating from order components that suddenly stop at the maximum rpm. Cursors have been inserted at several frequencies that were identified as poles of the system by applying the classical OMA. Moreover, the two gearboxes on the test rig are different and rotate at slightly different rotational speed, resulting in a doubling of those “end-of-orders” related identified poles.

This is the weakness of the method for which not only the real poles are identified, but also the so-called “end-of-order” related poles which are physically not present in the system [15]. The four identified frequencies correspond to some of the main order components ending at that frequency. The estimated modal model is not correct because it considers them as poles of the system.

3.2.2 Need for OBMA

The presence of the “end-of-order” related poles is the main reason for which a new method needed to be developed. The method described in section 2.2 does not suffer from this problem and only identifies physical poles of the system. It combines advanced Order Tracking techniques with Operational Modal Analysis to identify the resonances.

A clean tachometer signal is needed in order to have good orders estimates. The three shafts on the test rig (two high-speed shafts and one low-speed shaft) were instrumented by using zebra tapes combined with Keyence FU-10 probes. The rotating speeds can be measured by timing the duration of the passage of the alternating light and dark stripes glued to the shafts [16]. The main advantage of this system is that it is less expensive than other existing systems and it does not require shaft modifications or time consuming installations. On the other side, zebra tapes suffer from the fact that the last stripe has a different width than the others causing an erroneous signal that disturbs the analysis of the data. In [16] a butt joint algorithm has been developed in order to deal with fast rotating shafts.

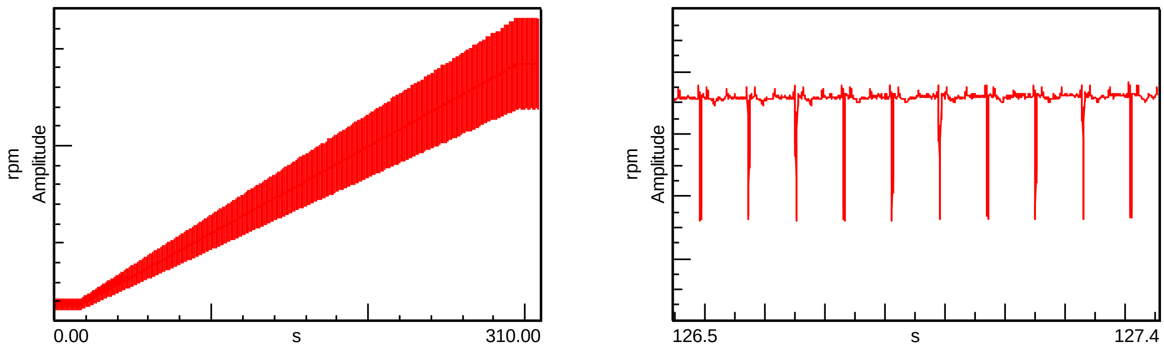

Figure 3: Tachometer signal before the application of the butt joint correction algorithm (left) and zoomed version within one second (right)

The algorithm is able to identify the location and the angular interval at the butt joint of a zebra tape from the tacho moments measured on a rotating shaft. This information allows the reconstruction of the exact angle evolution to be used for order tracking purposes. The algorithm makes use of an angle estimator function and a dedicated spline interpolation and FIR band-pass filter.

Figure 3 shows the rotational speed obtained from the zebra tape measurements when no butt joint correction is applied. On the left side the acquired signal during a measurement batch is shown, while on the right side there is a zoom within one second of test. The angle discontinuities clearly lead to spikes in the rpm data, making an order analysis impossible. Figure 4 shows the rotational speed after the application of the butt joint correction algorithm (right), where the spikes are clearly removed. The same figure shows also the zebra tape installation on one of the three shafts (left).

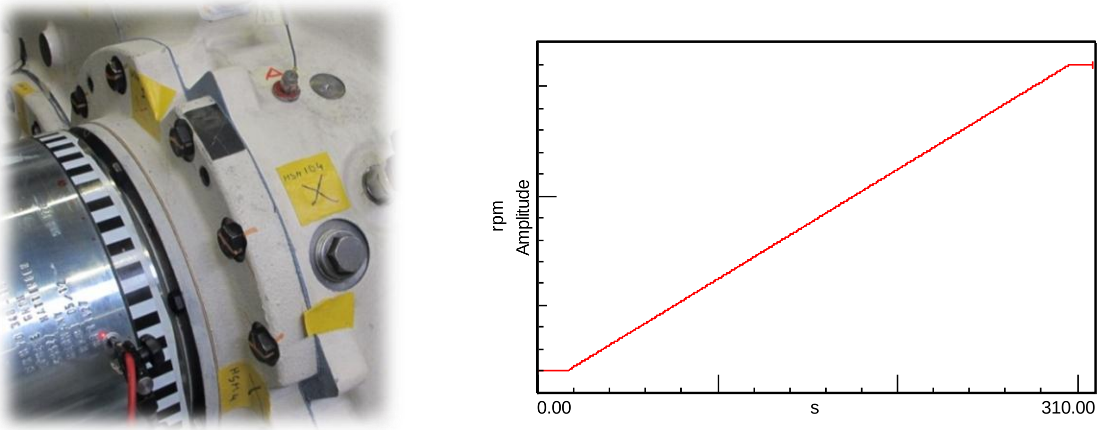

Figure 4: Zebra tape installation (left) and tachometer signal after the application of the butt joint correction algorithm (right)

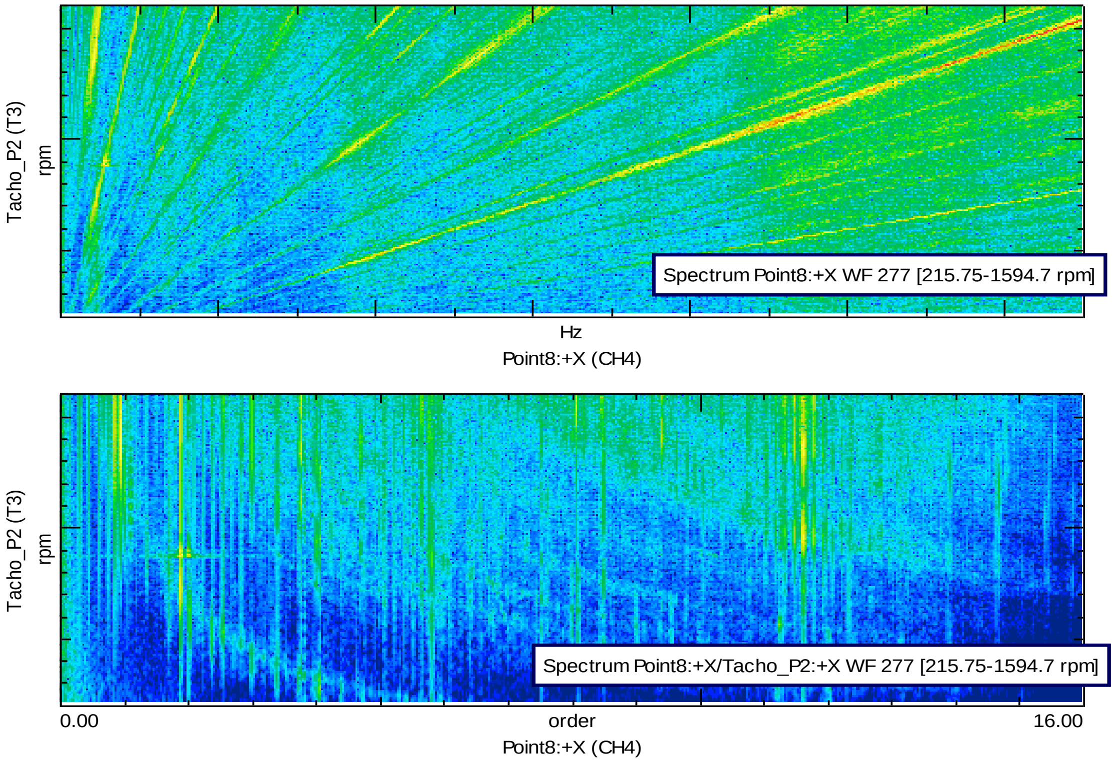

Figure 5: FFT spectrum map (top) and order spectrum map (bottom) for the same measurement point on the test rig

The Fast Fourier Transform (FFT) is used to transform the time domain data to the frequency domain. Signals that are periodic in the time domain appear as peaks in the frequency domain. To deal with orders, time domain data are converted to the angle domain to obtain samples at constant angular increments using the tachometer information as reference. Standard FFT algorithms are then applied to these data and, signals that are periodic in the angle domain appear as peaks in the order domain. For example, if a vibration peak occurs twice every revolution at the same shaft position, a peak appears at the second order in the order spectrum.

The most important difference between the two maps is how the peaks line up. Figure 5 shows the two different spectrograms for the same measurement point in order to underline the differences between them. The amplitudes of the peaks are indicated by the color in both maps. Each line of peaks on the order map (vertical line) clearly indicates a relationship between the shaft position and the vibration; the peaks in the spectrum map are difficult to relate to shaft speed. From both curves we can highlight the most interesting order that corresponds to order 11.63, which is the 2nd gear mesh frequency of the Intermediate Speed Stage (ISS).

In a first step, we can examine in details this order ignoring all the others. This is called order tracking. The order resolution has been set to 0.02 for a maximum order equal to 16, resulting in 800 order lines.

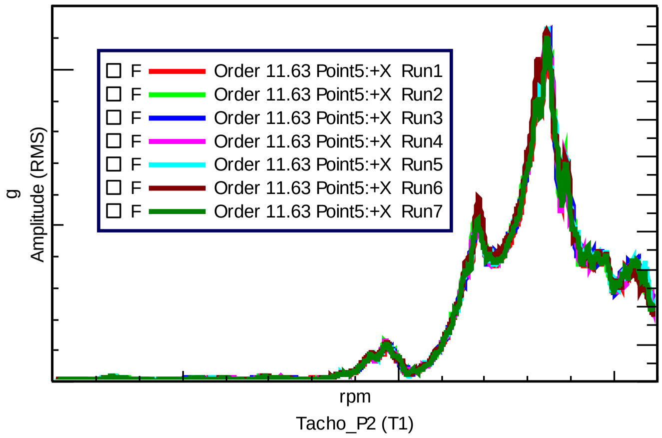

Figure 6: Most interesting order extracted from all the seven different data batches as a repeatability check

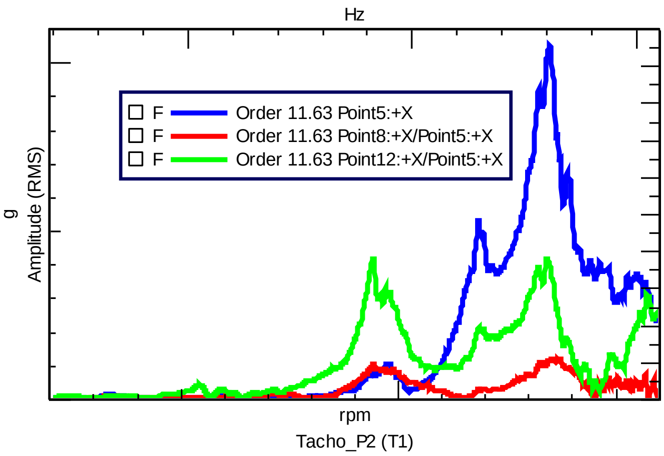

Figure 7: Most interesting order extracted for different measurement points during the same run

For standard analysis (order cuts, order levels) the focus is usually on the estimated amplitudes. In case of Order Based Modal Analysis, spectra need to be computed with respect to a reference signal. In fact, phase of all the points needs to be consistent; otherwise the assumed modal model will not be valid resulting in a faulty synthesized model. A standard way of selecting the phase reference is to choose one of the measured channels. Phases of all the other points will be made relative to this one. However a more robust and efficient method is to use a signal which gives a reference from the rotating source directly. Such a signal is the pulse signal from the tachometer. Since seven runs have been performed to cover all measurement points, a repeatability check needed to be performed. The same order has been extracted for a common point (point 5) and the results (see Figure 6) indicate a consistent data set

Figure 7 shows the extracted order for different measurement points during the same run. Even though some peaks can already be identified by only looking at the orders, Operational Polymax has been applied to these orders to extract the modal parameters. The stabilization diagram is quite clear and is shown in Figure 8. The vertical lines made up of ‘s’ identify the stable physical poles of the system.

By comparing the results with the ones obtained by applying the classical OMA, it can be seen that several "end-of-order" peaks are not anymore identified as physical poles of the system. Table 2 shows the comparison. The bold frequencies are the "end-of-order" poles identified by applying classical OMA.

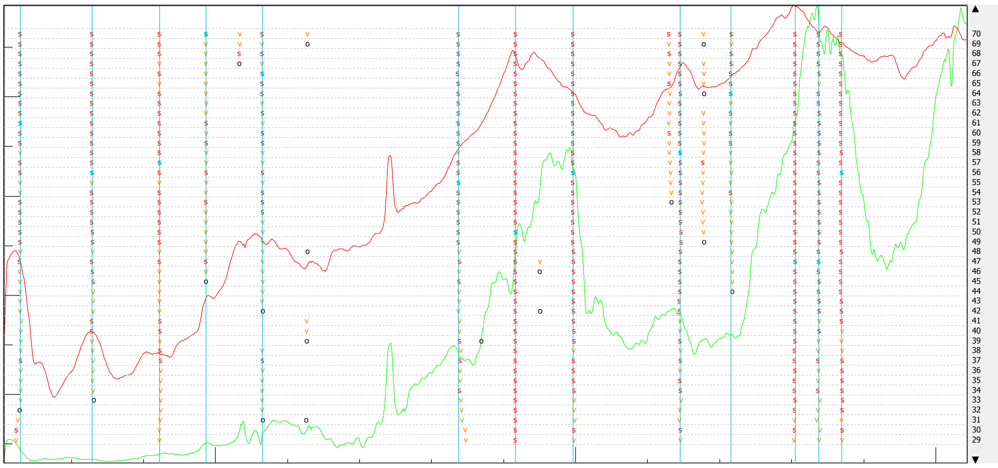

Figure 8: Stabilization diagram

Order Based Modal Analysis [Hz]	Operational Modal Analysis [Hz]
[40-50]	-
[60-70]	-
[80-90]	[80-90]
[90-100]	-
-	**[90-100]**
-	**[100-110]**
[110-120]	-
[160-170]	-
[180-190]	[180-190]
[190-200]	[190-200]
-	**[210-220]**
[220-230]	[220-230]
[240-250]	-
-	**[250-260]**
[260-270]	[260-270]
[260-270]	[260-270]
[270-280]	-
-	**[300-310]**

Table 2: Comparison between Order Based Modal Analysis results with Operational Modal Analysis results in terms of natural frequencies

Finally, in order to validate the identified modal model, the poles are plotted in Figure 9 on top of the rpm-frequency spectrogram to verify that they are not corresponding to "end-of-orders", as for the standard Operational Polymax case. It is quite clear that the proposed method does not suffer from "end-of-order" related problems identifying only the physical poles of the system.

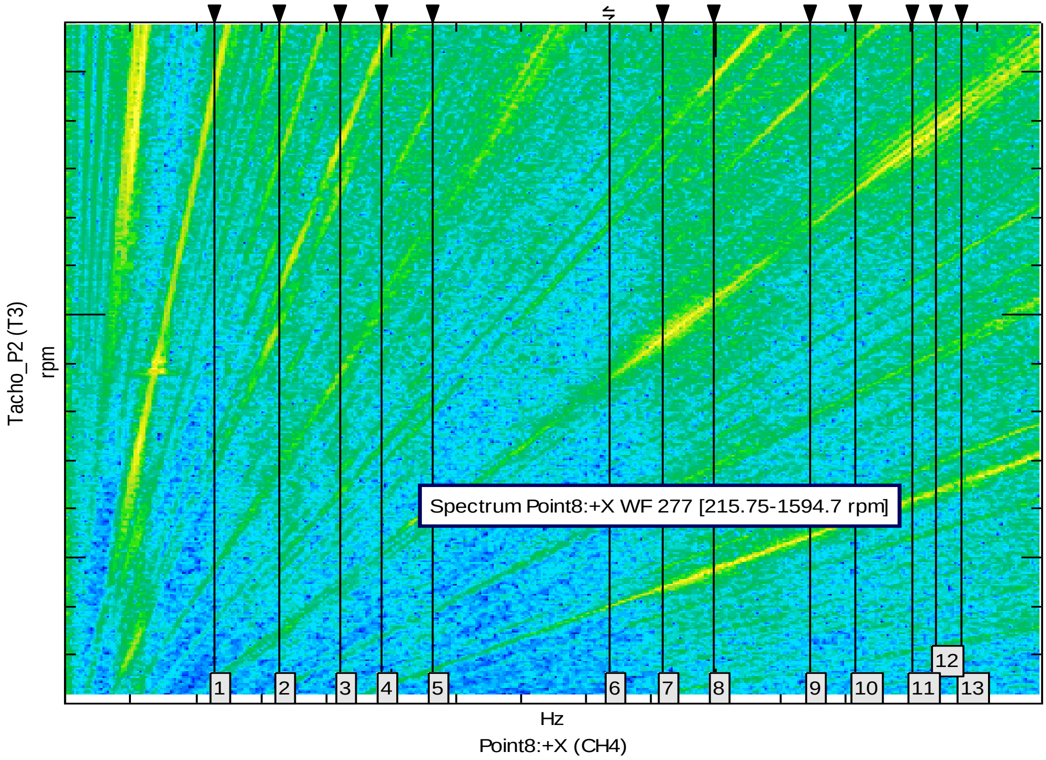

Figure 9: Rpm-frequency spectrogram of the gearbox run up in a certain frequency range and identified natural frequencies

4 Conclusions

The Order-Based Modal Analysis procedure has proven to be a promising technique for identifying the correct modal model in case of rotating machinery. The method has already been applied in the past to a car engine and to industrial pump systems, but here it has been tested on a very challenging case. The analyzed test rig configuration has two gearboxes with a slightly different gear ratio, and each one of them has several elements rotating at different rotational speeds. Several issues have been taken into account such as the butt-joint effect introduced by the zebra tape used to measure the rotational speeds, the noise in the measurements and the choice of the best suitable phase reference signal for tracking the orders both in amplitude and phase. The method performs better than the classical Operational Modal Analysis which suffers from "end-of-order" related peaks in the spectrum. By tracking the most significant orders and applying the Polymax parameter estimation algorithm to these orders, only the physical poles of the system are identified. In future work, the experimental modal model identified with this procedure will be correlated with an accurate flexible multi body simulation model of the test rig

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